REPLACEMENT OF THE BOILER FEEDPUMPS DURING THE RETROFIT OF THE 500 MW UNITS AT A GERMAN POWER STATION

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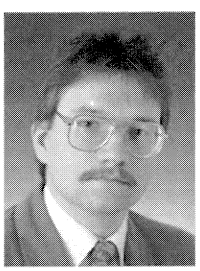

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ABSTRACT

The Jänschwalde Power Station consists of six 500 MW units that were commissioned between 1981 and 1989. These units, which were equipped with turbo feedpumps manufactured in Russia, had an average efficiency of only 78 percent and a mean time between overhauls of 25,000 hr. The task was to adapt these boiler feedpumps at the lowest possible cost (only minor modifications to the base and the pipework were envisaged), and to achieve a payback time of four to five years by improving the pump efficiency and reducing operating costs.

This was accomplished by fitting new cartridges into the existing barrel casings. These cartridges consisted of six-stages of a well-proven hydraulic with a specific speed of 1700 operating at the nominated turbine speed of 5600 to 5700 rpm.

The processing of the contract within the pump manufacturer's organization is described, along with the extensive coordination and quality assurance within a tight production schedule (delivery time of only seven months before the trial operation of the first pump unit). The efficiency of the new pumps was determined onsite by means of thermodynamic measurement. The specific features of the pumps and the proof of efficiency in the power station by means of thermodynamic measurement are defined in detail.

INTRODUCTION

Jänschwalde Power Station is located in the Federal German state of Brandenburg. It is a base-load plant with six 500 MW units designed for the firing of lignite. The units are of similar design and were commissioned between 1981 and 1989. The power station concept is based on the principle that two steam generators serve one 500 MW turbine, which is supplied by separate turbo feedpumps, in duo operation (Figure 1).

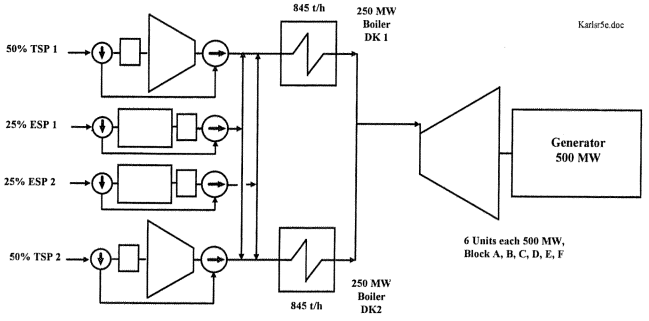


Figure 1. Pump Arrangement in the 500 MW Power Station at Jänschwalde.

In 1991, the utility decided to upgrade the units and retrofit flue gas desulphurization systems. It was also linked with the requirement to comply with the regulations for large-scale firing systems in the period from December 1994 to June 1996.

Five further measures, which were aimed at improving the efficiency and the availability of these units, were also undertaken [1, 2]. These investments created the preconditions that will enable the power station to be operated more efficiently for another 20 years.

The retrofitting of the turbo feedpumps was also investigated.

STATE OF THE FEEDPUMPS

(DRAWBACKS OF THE OLD DESIGN)

All the units were equipped with PT 850 turbo feedpumps of barrel casing design. Although the two 50 percent turbo feedpumps for each unit were safeguarded by a parallel connection of two 25 percent electric feedpumps as standby (Figure 1), retrofitting of the turbo feedpumps was considered especially important because of the long average running time of 7500 operating hr per year. In addition to a:

- Marked increase in efficiency
- Lengthening of the overhaul periods for these pumps
- Assured supply of spares
- Possible replacement of the inner casing within a period of 72 hours

it was desirable to increase the availability and, by means of goal-oriented monitoring, to change over to actual state maintenance.

An analysis of the hitherto unplanned stoppages revealed the following weaknesses of the existing feedpumps (principal design features, Table 1, Column 1):

- Design of the axial thrust balancing device in the form of a disk (Figure 2, excessive wear during transient operation).
- Seal water supply system on the throttling bush shaft seal (Figure 3, high heat losses, complicated system, shaft prone to bending during transients).
- Thermal problems in connection with the shaft seal (consequently lowering of the bearing support, increased wearing clearances, likely rotor instability).
- Poor fit of the inner and outer casings (following several cartridge changes).

With an average period of 20,000 to 25,000 operating hr between overhauls, the utility had to bear costs of about $150,000 for each overhaul. A further obstacle was that, for various reasons, it became increasingly difficult to obtain original spares from the pump manufacturer.

Table 1. Data of the Existing and the Retrofitted Turbo-Boiler Feedpumps.

	1	2
	Initial State	Retrofitted Variant C (Execution)
BOOSTER PUMP		
Type	HGD-300/2/40 A	HZB 253-640
No. of stages	2	1
No. of flows	single flow	double flow
Shaft seals	Stuffing box	Mechanical seals
Gear ratio (gear to booster)	1 : 4	Existing
MAIN PUMP		
Type	PT 850-250	HPT 300-315-6s/33A
Design	Barrel	Barrel
Cartridge	In series with horizontally split case	In series with radially split case
Diffuser	4-vaned diffuser	Diffuser with. closed channels
Balancing device	Disc	Piston
Shaft seals	Fixed throttling bushes	Mechanical seals
Bearing span	7.8 ft (2380 mm)	7.3 ft (2222 mm)
Barrel OD.	45.2 in (1150 mm)	Existing
Barrel ID.	28.5 in (725 mm)	Existing
Wear Ring Clearance	170%	100%
Impeller Diameter D_2	100 %	94 %
Weight of the barrel	11000 lbs (5000 kg)	Existing
No. of stages	5	6
Specific speed nq	28.8	33
NS_{usa}	1500	1700

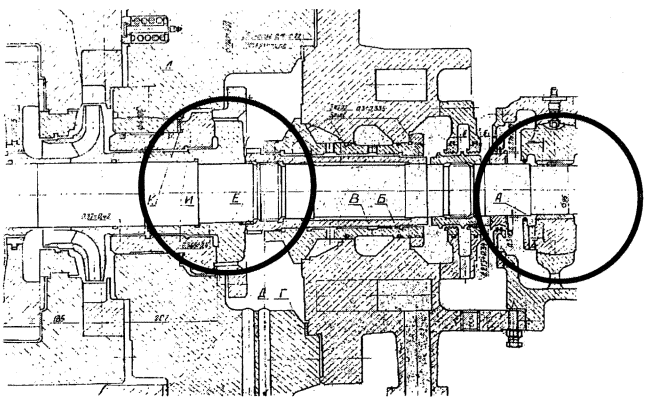


Figure 2. Technical Problems of the Pump Built in Russia.

In view of the aforementioned reasons, the utility decided to investigate various possibilities for the retrofitting of these pumps.

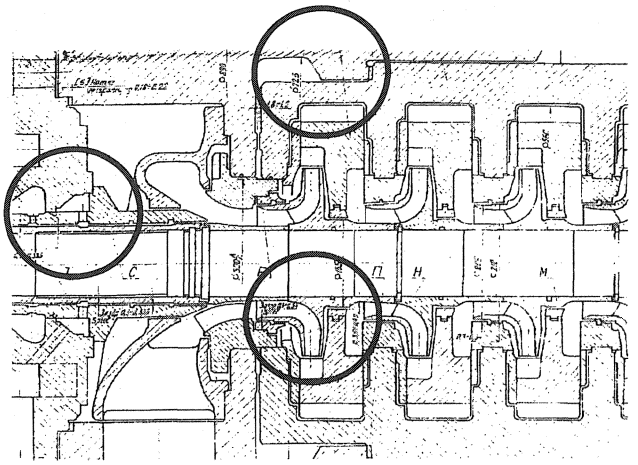


Figure 3. Technical Problems of the Pump Built in Russia (Continued).

The objective thereby was not to change the existing suction and delivery pipework or the foundations, but to adapt the connecting lines, base frame, and supply lines at a minimum cost.

ECONOMIC CONSIDERATIONS

Three fundamental variants have resulted from the given marginal conditions:

- Retrofitting of the existing PT 850 main pump and the employment of a new booster pump.
- Replacement of the booster and main pump with the latest machine types produced by the respective manufacturer and adaptation to the existing pipework, foundations, and base frames, etc.
- Employment of a new cartridge with improved hydraulics, further use of the existing barrel casing and adaptation to the new cartridge (thus, utilization of the existing pipework connections along with the base frame of the main pump), and also adaptation of the pipework connecting the booster and main pump.

The respective interpretations are compared in Table 2 of the three variants of the initial situation according to their relevant design/manufacture and economic aspects. As shown in this table, there is no further apparent, decisive technical advantage between the Variants B and C. However, the significant role that the operating and additional costs (maintenance costs) play in this contract is clearly evident. Based on the assessment criteria laid down by the utility, only the Variants B and C were pursued further (Table 2).

Table 2. Possibilities of Retrofitting an Existing Boiler Feedpump.

			Initial state	Variant A	Variant B	Variant C
			Existing pump PT 850 (old state)	Existing pump PT 850 (retrofitted)	Ideal choice Sulzer (latest develop.)	Tender to VEAG Old barrel+ new hydraulic
PREREQUISITES	Flow Qd	US GPM	15,000	15,000	15,000	15,000
	Head H	(FT)	9,030	9,030	9,030	9,030
	Speed of Turbine	(RPM)			5600-5700	5600-5700
DESIGN						
Booster	No. of stages		2stages / single flow	1 stage / double suction	1 stage / double suction	1 stage / double suction
	Head	(FT)	669	433	433	433
	Speed	(RPM)	1450	1460	1417	1417
	η Booster	(%)	82	85	85	85
	D2	(IN)	22.6	22.6	25.2	25.2
Main Pump	No. of stages	(-)	5	5	6	6
	Head	(FT)	8330	8580	8590	8590
	Speed	(RPM)	5900	5840	5666	5676
	η Main Pump	(%)	78	80	85.3	85 (guarantee)
	Impeller Dia. D2	(IN)	13.2	13.2	12.4	12.4
	Shaft seals		Fixed throttl. bushs	Mech. seals	Mech. seals	Mech. seals
	Bearing span	(IN)	87.5	87.5	78.7	87.5
	Clearance sD / nominal cl.	(%)	170%	146%	100%	100%
	Weight	(LBS)	24,670	24,670	15,420	24,230
ECONOMY						
First costs	Booster- + M.-Pp.	(%)	-	30	75	60
	Adaptations	(%)	-	-	18	8
Variable costs	Operating costs	(%)	388	270	-	12
	Maintenance costs	(%)	100	50	20	20
Price for comparison *)	Total costs	(%)	488	350	113	100%

*) related to 15 years

For evaluation of the existing offers, giving due consideration to the specific dependencies of the power station, the following aspects are of the greatest importance for the profitability statement:

- Investment costs for the new feedpumps
- Residual life of the power station installation and the number of operating hours per year
- Saving of driver power (converted to the savings on coal)
- Overhaul and maintenance costs
- Time point for the replacement of the pumps (e.g., in connection with the scheduled stoppages)
- Consideration of the extra costs associated with the operation of electric feedpumps

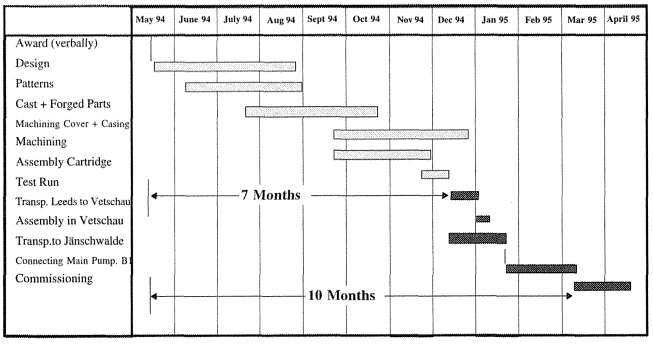
With the most favorable variant, the profitability statement of the utility showed a payback period of four to five years. On the basis of this, the utility decided in April 1994 to implement this retrofit measure as well.

CONDITIONS LAID DOWN BY THE UTILITY FOR THE PLACEMENT OF CONTRACT

Prior to concluding final negotiations, the minimum technical requirements, scope of guarantees, along with proof of the proposed characteristics were defined between the partners. They embodied the following main points:

- Date of the replacement–It was the aim of the utility to link the stoppages of the units with the flue gas desulphurization systems, and thus also to install the retrofitted feedpumps. This meant a delivery time of only 10 months to the commissioning of the first feedpumps and further intervals of two months in each case for the delivery of another two booster and main pumps (Table 3).

Table 3. Time Schedule for the Manufacture of the First Pump (B1).



- Assurances for the retrofits–Since there would not be sufficient time to dismantle and refit the feedpumps, in order to refurbish the same barrel casings, two barrel casings would have to be available in advance in each case.
- Technical requirements–To ensure the economic viability, an efficiency greater than 84 percent and an output time of greater than 50,000 hr had to be guaranteed. Furthermore, according to VDI 2056, Group T, (vibrations of bearing support) a vibrational behavior in the category "good" was required for the load range of 40 to 100 percent.
- Guarantee requirement for the further use of the barrel casing–Further use of the barrel casing was only acceptable to the utility, if:
  - Mathematical proof was provided for a residual life of 20 years.

• Quality assurance steps for material and weld examinations were carried out in such detail that they would suffice as proof for all the relevant positions (size of defects and maximum number of indications were to be noted exactly for every critical point).

• A new component was to be delivered if a casing or pressure cover was rejected.

• Proof of the proposed characteristics–Thermodynamic acceptance measurement under operating conditions was required as proof of efficiency. Since this efficiency and, hence, the resultant operating costs were of particular importance for this contract, the efficiency tolerance according to DIN 1944 / I was limited further to the maximum accuracy of the measuring device (i.e., to maximum minus one percent). A penalty of $3700/kW would be payable if the efficiency was not achieved. With this procedure, all specialist publications concerning the use of mathematical models for the determination of the hot water efficiency were regarded as unacceptable.

SELECTION OF PUMPS AND PUMP PRODUCTION LOGISTIC

In view of the required efficiency, the specified requirements and the price advantage (Table 2, Variant C) in comparison with a completely new pump (Variant B), the pump manufacturer offered a six-stage cartridge in the existing casing with corresponding adaptations (Figure 4). Although the original pump was of five-stage design, a six-stage cartridge could still be fitted in the existing barrel casing (Table 1, Column 2).

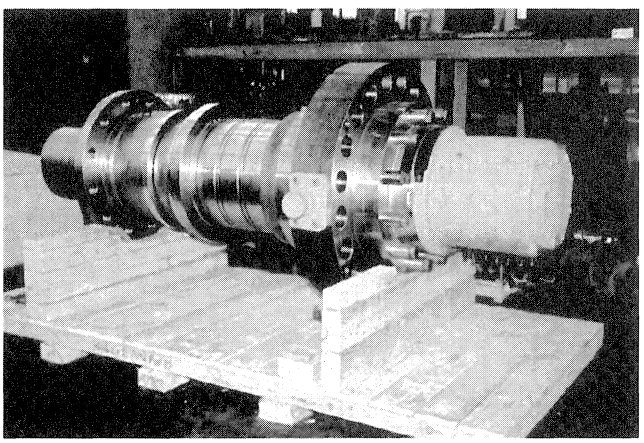


Figure 4. Cartridge of the Boiler Feedpump HPT 300-315-6s/33A at Jänschwalde.

A hydraulic with a specific speed of NSus = 1700 (≈ nq 33) could be employed for the six-stage pumps. It has acquired an excellent reputation in more than 100 executed installations.

The hydraulics with a higher specific speed than that of the original design was used to increase the efficiency.

The pump manufacturer was awarded the contract for the retrofit of the 12 turbo feedpumps in May 1994. The conversion of the mentioned turbo feedpumps was entrusted to the German Division and may be quoted as an example of international cooperation with the respective advantages and disadvantages. The German Division was responsible for the order processing and the commercial aspects of the contract. Since the parent company in Switzerland is responsible for the design and construction of large feedpumps, the draft design work was made in this country. Due to the pressure of time and capacity, the pump cartridges were manufactured, fully assembled, and works tested by the division in Great Britain. Subsequently, they were shipped to the power plant maintenance facility in Germany. There adaptations were made to fit the cartridges to the casing already in place. This is a typical example of today's globalization of order processing in Europe.

NEW DESIGN FEATURES OF THE REPLACED PUMPS

The new booster pumps, type HZB 253-640, are built as single-stage radially split machines with a double flow impeller of proven design in order to further improve the suction conditions. In contrast to the former booster pumps with packing-type stuffing boxes, mechanical seals are now employed (Table 1). The internal mechanical seal-water circuit is equipped with two magnetic filters and an intensive cooler, which is integrated around the shaft, on each pump side.

The main pump, type HPT300-315-6s/33A, is a centrifugal machine with barrel casing design. The suction and delivery branches are arranged vertically facing downwards (Figure 5). The bearing span of 7.28 ft is governed by the existing casing of the pump, i.e., about 10 percent more than the span usually needed for a six-stage hydraulic. From the standpoint of the bearing span, the selected six-stage pump corresponds to a seven-stage machine (Figure 6), when it is manufactured according to the latest principles. Consequently, the static sag of the retrofitted pump rotor is greater than that expected with a brand new machine.

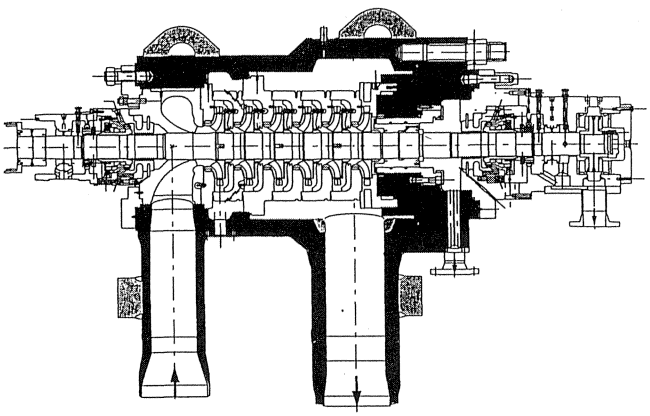


Figure 5. Cross Section of the Pump.

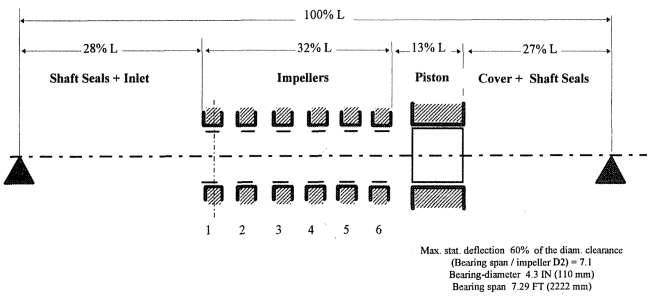


Figure 6. Bearing Span at Jänschwalde (New Cartridge in Existing Casing).

Since the existing casing had been in service for some 15 years, it had to be established, as already stated by means of a residual life analysis–whether or not the fatigue strength of the casing had been damaged by the full load-temperature cycles experienced hitherto in service. The maximum stress occurs at the connecting point of the delivery branch. With the aid of Wöhler curves for this material, the effect on the fatigue strength of the material was determined from the peak stress resulting from the internal pressure, pipework load, and the temperature transients. The

analysis indicated that the reduction in fatigue strength was only four percent of the permissible value. In other words, the required life cycle had been hardly affected by the previous operation of the pump.

With the aid of an appropriate assembly device, the cartridge design, i.e., with internal block consisting of rotor, inner casing, delivery cover, balancing device, shaft seals, radial and axial bearings, facilitates rapid replacement and, thus, reduces downtime during service appreciably.

The impeller and diffuser were precision-cast in the most accurately possible shape and with a smooth surface in order to obtain optimum efficiency. The impellers are shrink fitted to the shaft to guarantee good concentricity and assure sealing of the stage pressure. The hydraulic axial thrust is taken up by a shaft shoulder on the first stage and by split-rings on the following stages. The torque transmission is achieved by two opposite keys on each impeller.

The axial thrust is largely compensated by means of a cylindrical, nonstepped balance drum. Thus, no touching during transients can occur. The residual thrust is taken up by a tilting pad axial bearing that can be subjected to load on both sides. This ensures sufficient safety, even with maximum running clearances and increased coupling thrust. The piston is shrunk-fitted to the shaft. The piston can be easily dismantled with an appropriate hydraulic device.

Swirl brakes are employed in the balancing bushing of the balance drum. These ensure good damping of the pump rotor in the event of clearance wear. Further design features and information concerning the material were described already in Burchhardt, et al. [3].

A mechanical shaft seal, type 270 F 6.5 in, with a material combination silicon carbide paired with synthetic-impregnated carbon, has been fitted as shaft seal. The circulation is affected via an externally located cooler and magnetic filter, and without any dosing circuit as is customary for an operation in combined oxygen treatment (feedwater between alkaline and neutral). A cooling chamber is provided to reduce the heat transfer from the hot pump in the direction of the rotating seal ring. The mechanical seals can be dismantled as a complete unit. During operating conditions, the circumferential speed in the middle of the rotating seal ring is about 157 ft/s. The leakage quantity measured in the installation is less than one liter per hour.

Following their removal and dismantling, the casing and cover of the old main pump were examined for possible further use, remachined for the fitting of the new cartridge and then subjected to another quality inspection. The positions that had to be adapted for the new cartridge are shown in Figure 5.

PRACTICAL EXPERIENCE DURING THE REPLACEMENT OF THE PUMPS

For the conversion of each pump group, there was only a short time window available for the installation of the pumps. This was linked with, and determined primarily by, the respective flue gas desulphurization system. Any delay in meeting the deadline was subject to a heavy penalty of 0.1 percent of the contract value per calendar day.

The first pump group had to be ready for operation in the plant on March 1, 1995. For the design, models, castings, manufacture, and the erection at works, there were only seven months before the trial run at works (Table 3). Quick decisions were therefore required, and needless to say, several of them were based on assumptions. Unfortunately, it transpired that some of those assumptions were incorrect.

Two premanufactured barrel casings were always available as forerunners. As is normally the case, it was assumed that all casings and covers were identical and interchangeable. However, it was established in the processing of the contract that only one cover matched the respective casing in each case. When it came to erection, it was clear that the holes in the cover did not match up with the bolt pitch circle (Figure 7).

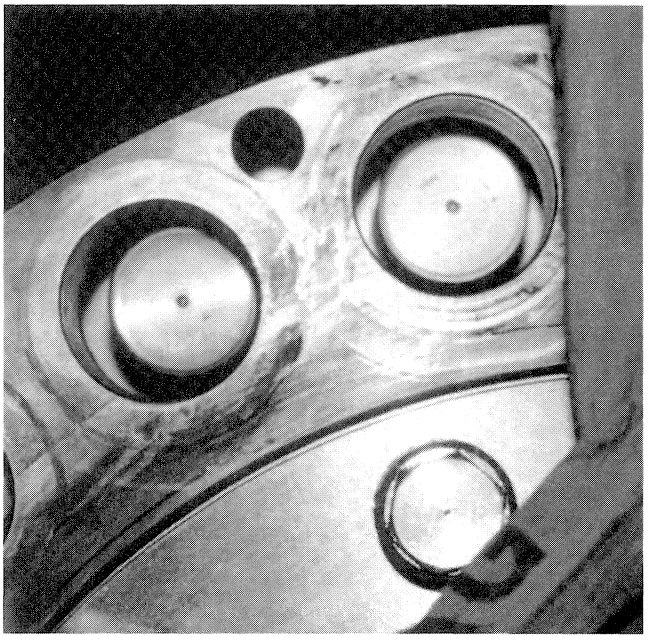


Figure 7. Dislocation of the Cover Bolts at the Existing Barrel.

Adaptations had to be made which were not planned in the time schedule. And since the time window could not be changed to provide for the additional work, this increased the pressure of time even more. It necessitated exceptional efforts by the work team in the service center.

Different casings had to be repaired, because the ultrasonic and liquid penetrant tests revealed cracks and porous lines in the welds. The cracks were ground out and welded. With one casing, the delivery branch had to be cut off and welded again.

OPERATING EXPERIENCE

The commissioning of the first pump was completed on schedule. The pump ran very smoothly (bearing support vibration less than 0.08 in/s, together with "good" vibrational behavior in the operating conditions from minimum flow to maximum load, i.e., from 25 percent up to approximately 100 percent best efficiency point flow).

After the commissioning of the first three pumps, however, Pump A2 suffered damage. During the commissioning and the functional testing of the mechanical overspeed safety device of the drive turbine, which is effected in Jänschwalde in the coupled condition with the pump, Pump A2 seized up during the rundown. This created a critical situation in the processing of the contract, because the cause of damage was not readily apparent.

The pump was dismantled and scuffing marks were found on the sealing rings of two impellers. An analysis of the cause of damage showed that instead of the proven impeller material G-X5CrNi13.4, the material 17/4 PH (≈G-X7CrNiMoNb 15.5 with 17.5 percent Cr and 7 percent Ni) had been substituted for reasons of expediency. This material was paired with the material 1.4138 wnt (soft nitrated)(≈ASTM A743 Gr.CC-50), which was obtained locally and did not meet the original specifications. In addition, it was presumed that the scuffing marks with Pump A2 had occurred already during inhouse tests on the test stand. A strip test had been dispensed with because of the tight time schedule.

To enable the pumps to be put quickly into operation again, it was decided to open the clearance of the impeller sealing rings by 20 percent. Since all the supplied pumps had been manufactured with the same material pairing, all the subsequent machines were provided with the corresponding opened clearances. In addition to the larger gap clearances, smooth impeller gaps were also employed in these special cases. Extremely careful erection of the inner cartridge facilitated further operation with the above mentioned material pairings.

The larger bearing span conditioned by the use of the existing casing naturally resulted in increased static sag of the shaft, which favored the scuffing. Precise alignment of the impeller in the casing is therefore of decisive importance.

The pump in operation is exceptionally quiet. The frequency spectrum is shown in Figure 8 and Figure 9 of the vibration of the Pump F1 at the continuous operating point in the plant. This pump already has an opened impeller gap clearance. The harmonics of the speed frequency are designated in Figure 8 and Figure 9 with 1×, 2×, etc.

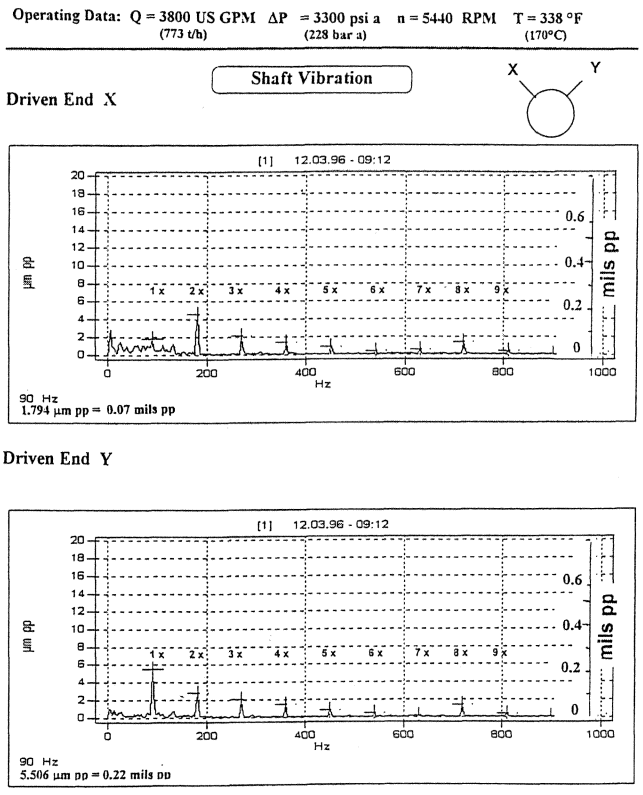


Figure 8. Frequency Plot of Shaft Vibration.

The following provide information about the rotordynamic behavior of the pump:

- Shaft vibration, driven end (Figure 10).

• Zero to 1×–There are no peaks, i.e., the rotor runs stable due to the geometry of the gaps. There is only slight broad band excitation, i.e., the turbulence is only weak (reason: good flow guidance).

• 1×–The peak is 1.8 µm p/p (in x-direction), respectively, 5.5 µm p/p (in y-direction) (filtered for rotational frequency) NB. This value corresponds to a cumulative value of Smax≈0.35 mils p (9.1 µm p) (unfiltered) and provides information about the sum

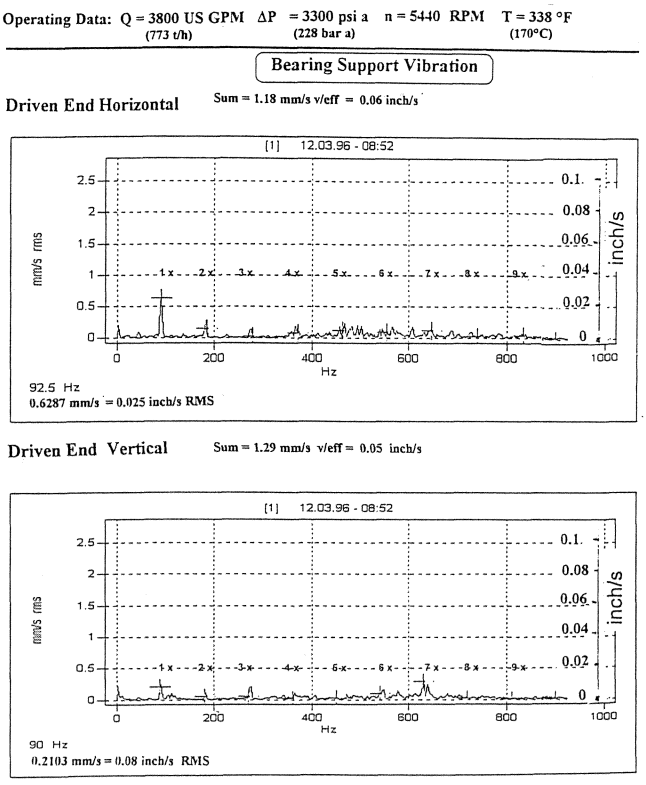


Figure 9. Frequency Plot of Bearing Support Vibration.

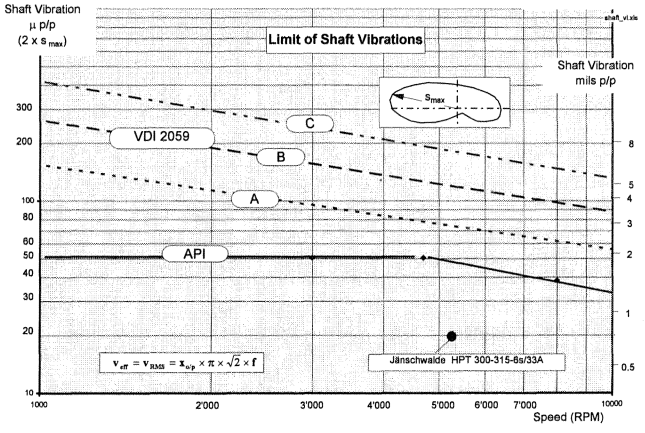


Figure 10. Limit of Shaft Vibration.

of the mechanical and hydraulic exciting forces. This value is very low. If this shaft vibration is compared with VDI 2059, Page 3, 1985, the value for good vibrational behavior is ≈1.25 mils p (32 µm p), and the measured value is, thus, appreciably lower (Figure 10).

• 2×–The peak is 4.5 µm p/p (in x-direction) resp. 2.7 µm p/p (in y direction), this value is influenced by the gear coupling.

• 7×–Is the impeller vane passing frequency (number of impeller blades multiplied by the rotational frequency). This value is extremely low.

- Bearing support vibrations, driven end as above (Figure 9).

• 1×–Peak / v eff = 0.024 in/s (horizontally), resp, v eff = 0.008 in/s (vertically) NB. The cumulative value is v eff = 0.046 in/s (horizontal) and v eff = 0.05 in/s (vertical). According to VDI

2056 Machine Group T 0.1 in/s is admissible for good vibrational behavior (Figure 10).

• 2×, 3×, etc.–The higher harmonics in the frequency spectrum exhibit only very low values.

The very low level of broadband vibrations is an indication that the flow guidance within the pump is at optimum. Other reasons include good casting quality, correct choice of the number of impeller and diffuser vanes and all critical dimensions of the impeller plus the clearances between impeller and side walls being within the specified tight tolerances. The high efficiency test achievement is a further reflection of the above mentioned features.

The guarantee values were realized in full. The smooth running of the pump along with that of the rotor are the best prerequisites for the operation of the pump resulting in low wear of the clearances, bearings, and mechanical seals and, therefore, only a slight decrease in efficiency is expected over the operating life of the pump.

SCALE-UP EFFECTS OF EFFICIENCY

The guaranteed efficiencies are based on the efficiencies of many manufactured pumps and a basic hydraulic, which was measured cold in a model machine at a speed of about 1500 rev/min. This included the measurement of the clearances, piston leakage, bearing, and seal losses. The disk friction of the impellers and the piston can be calculated sufficiently accurately. From these values, it is possible to determine the hydraulic efficiency. In the full-scale version, the hydraulic efficiency was scaled-up according to the Sulzer step-up formula [4, 5] for impeller diameter D2, temperature, and speed (Figure 11).

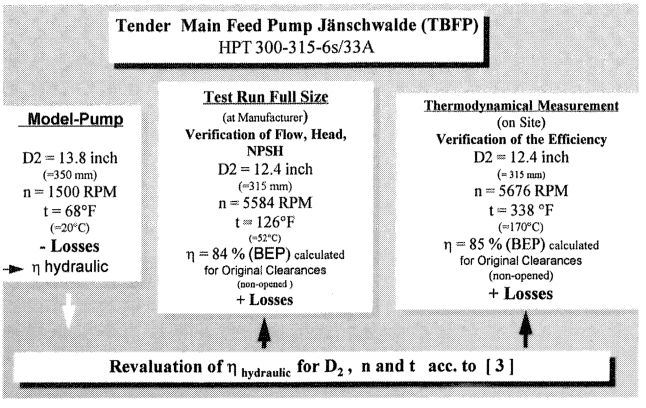


Figure 11. Method of Scale-Up Effects of Efficiency.

This means that the expected hydraulic efficiency of the full-scale version is known. With the clearances of the full-scale version, the bearing and shaft seal losses that were specified by the suppliers, it is possible to determine the attainable efficiency of the full-scale version. Since the pump operates at ≈ 95 percent of the best point quantity in the operating point, the operating efficiency is slightly lower. The guaranteed efficiency is 85 percent.

In principle, the hydraulic efficiency should always serve as the basis for all efficiency determination formulas. However, if the effective efficiency measured cold at reduced speed on the test stand is converted with the Karassik formula [6], for example, it usually results in excessively high values. This is quite understandable, because mechanical and hydraulic losses cannot be converted in the same manner. (The hydraulic efficiency hereby is the efficiency, which considers the efficiency of the stage, i.e., friction in the impeller and diffuser channels, inlet and outlet losses. It is recalculated by means of the volumetric, mechanical losses, and disc friction losses (Figure 11).

As shown in Figure 12, the distribution of the Jänschwalde feedpump efficiency, Nsus = 1700 (nq = 33) at the best point. (One hundred percent minus the sum of the depicted losses in efficiency points = the offered efficiency). It is, therefore, apparent that the hydraulic efficiency of the pump stage is responsible for the greater part. From this, it is also obvious that a good pump hydraulic can influence the overall efficiency the most. For this reason, expensive diffusers with closed channels and spatially curved overflow are employed here.

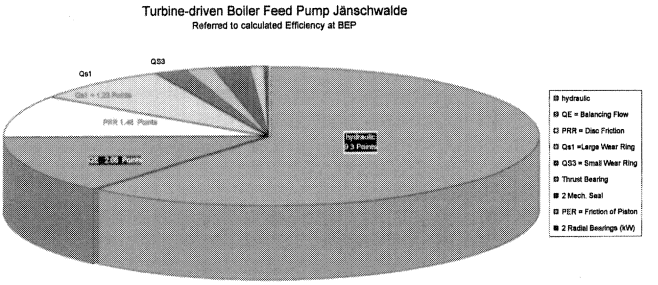


Figure 12. Allocation of the Losses of a Boiler Feedpump.

Also shown in Figure 12 is that the efficiency here is only dependent to the extent of 25 percent on the clearances in the labyrinths and the balancing system. The mechanical losses of the bearings and mechanical seal have a rather subordinate influence.

VERIFICATION OF THE EFFICIENCY (WORKS AND FIELD MEASUREMENT)

The new cartridge (No. 104) was tested in cold condition at full speed (125°F, 5676 rpm with normal clearances) on the test bed in Leeds, Great Britain, on December 12, 1994, just seven months after the order. An efficiency of 84.6 percent (cold) in the operating point was achieved at the first attempt. The delivery head was very close to the offered curve within the guaranteed limits. Likewise, the NPSH was also maintained exactly (Figure 13). The head with closed valve was slightly higher, but still within the guaranteed tolerance of three percent. The main aim of this test was not the determination of the efficiency, but the Q/H/NPSH characteristic of the pump.

To be safe, one pump was also tested in the works in hot condition (338°F) and full speed, because this is possible on the test stand in Leeds, Great Britain. At the same time, a hot water efficiency of 86.2 percent was determined with the customary measuring instrument tolerance (about two percent).

Each aggregate in the power station was further subjected to a separate acceptance measurement, by which each pump was jointly measured thermodynamically by specialists of the utility and the pump supplier. The aim of this test was the determination of the efficiency at operating conditions. The payment of a penalty is based on this efficiency. The principle of the thermodynamic measurement is depicted in Figure 14 using an example.

To obtain results of high accuracy, a recorded data collecting system was developed in cooperation with the research division. It enables continuous measured values to be called up, stored, and to select certain measurement points for the evaluation (Figure 15). The great advantage with this procedure is that all steps are reproducible and thus transparent for all concerned. The collection of the measurement data was made over a period of about three hr, during which all measurements were called up and stored at intervals of 14 seconds (verification according to DIN 1944 / I [7], whereby the inaccuracy of measurement for efficiency was limited to one percent). By plotting the values against the respective time periods, it was possible to see exactly when the steady-state conditions occurred (Table 4).

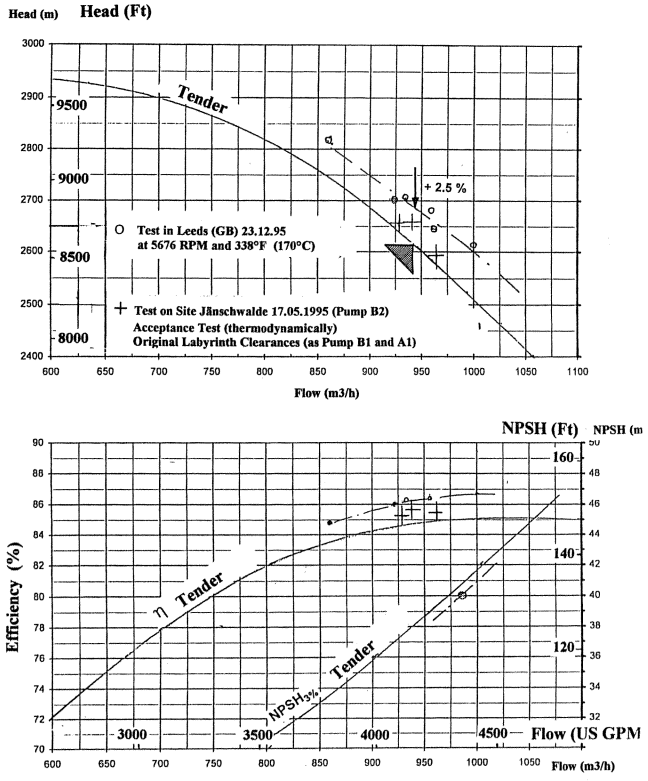


Figure 13. Comparison of the Calculated Characteristic vs *the Measured Characteristic.*

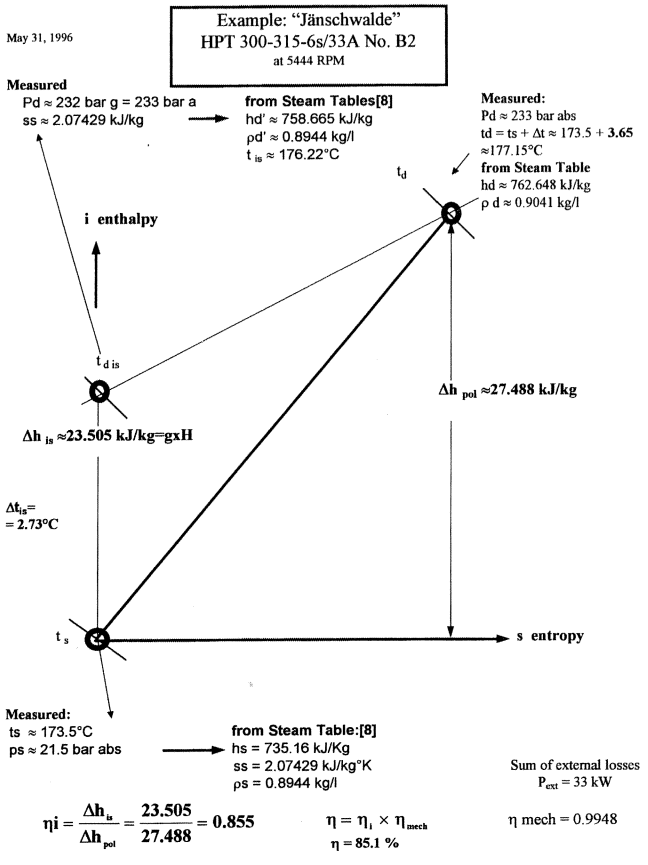
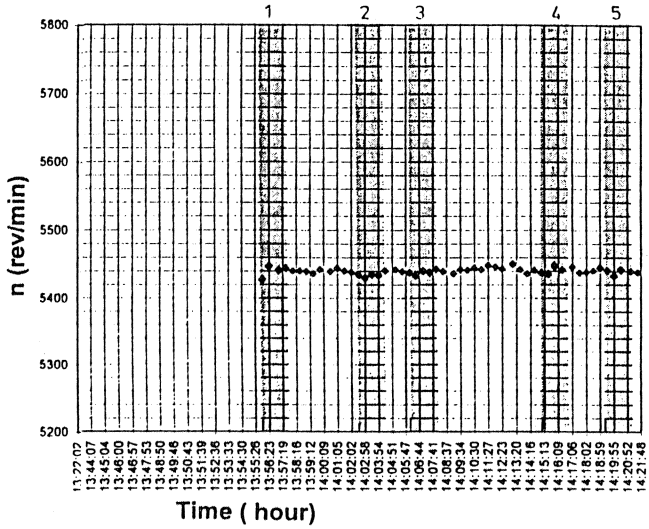
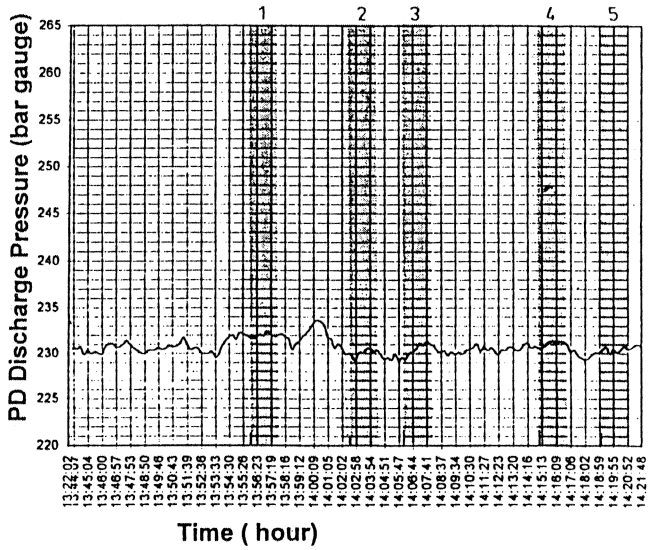


Figure 14. Thermodynamic Efficiency Calculation (Example).

Table 4. Stability of the Measurements According to the Thermodynamic Testing Method.





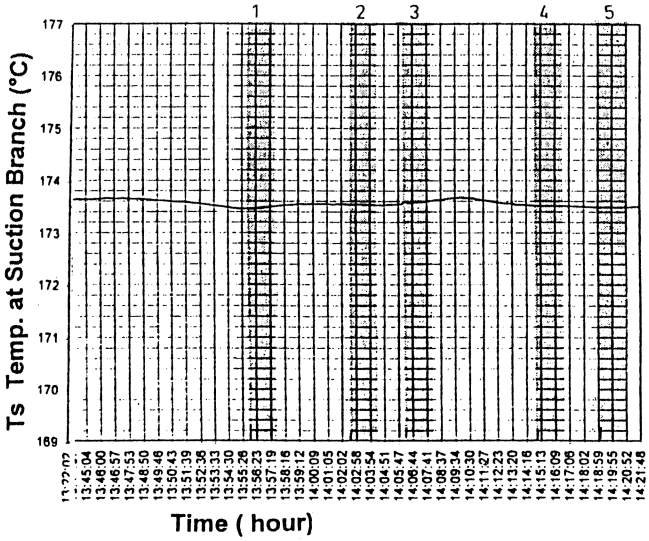
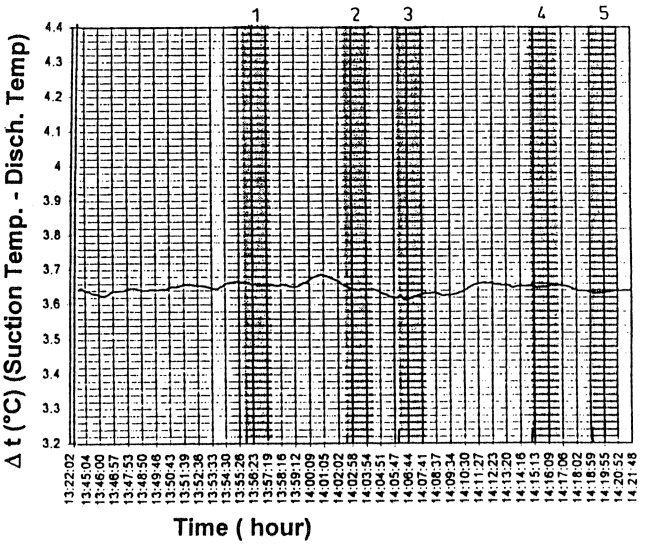


Table 4. (Cont'd)



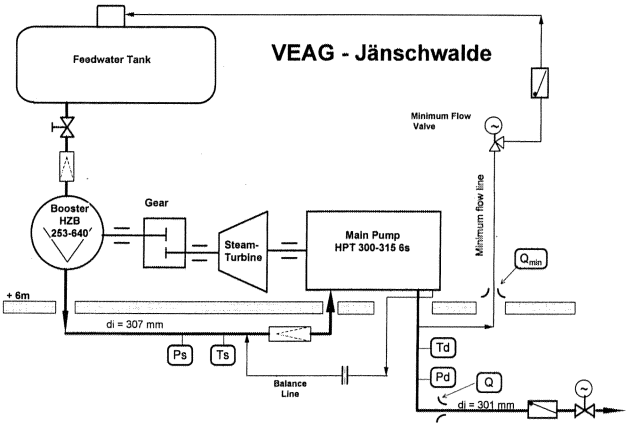


Figure 15. Position of the Instruments for the Thermodynamic Measurement Onsite.

The efficiency was calculated from the average of these values. Despite the opened clearances, the guaranteed efficiency was attained for all pumps by means of thermodynamic verification in the plant. It should be noted that the first three pumps (B1, B2, A1) are still operating with the original clearances and attaining an efficiency of about 85.6 percent, i.e., 0.6 percent above the offered efficiency of 85 percent. The measured pump efficiency plotted as a function of the wear ring clearance is shown in Figure 16. The efficiency of the original pump has been also measured by thermodynamic method and was only 78 percent, the measuring tolerance however was approximately ± two percent.

As of September 1996, the first new pump has run 10,000 hr. In order to detect a possible efficiency loss during this time, the pump A2 (with original clearances) was again measured by thermodynamic method by the experts of the utility. A decrease of only 2/10 of a percent in efficiency was detected, which lies in the range of the measuring tolerance.

If the weighted instrument and measuring tolerance are combined with the tolerances of the values of the steam table [8] (≈ 0.5 percent), a total inaccuracy of the efficiency-measurement with the thermodynamic method results in a tolerance of approximately ± one percent. Further details are available in the literature [9, 10].

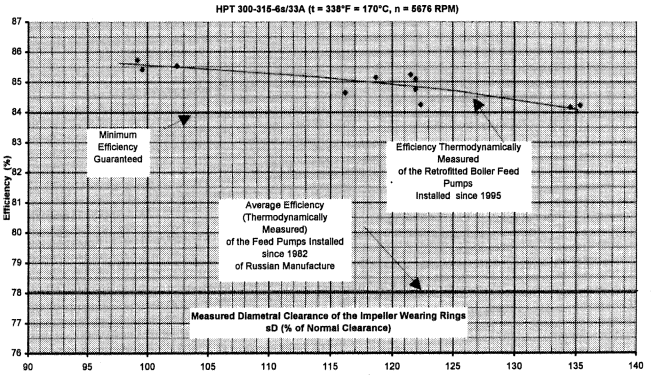


Figure 16. Thermodynamic Efficiency Measured Onsite.

CONCLUSIONS

The customer's requirement to retain the existing turbines, fundamental anchorage points, and pipework connections, was fulfilled by the pump supplier through the delivery of new cartridges for installation in the modified casings.

The allotted time for the upgrading of the turbo-driven pump units at Jänschwalde power station was not exceeded relative to the planned time window. The availability of the power station was not influenced by the retrofitting of the feedpumps.

The extremely short delivery time of only 10 months from the receipt of order to commissioning (Table 3) called for a high involvement of the working personnel. They had to be full of ideas to solve the problems that suddenly occurred time and time again at short notice. The cooperation of the European companies of the pump supplier is worthy of special note in this respect. All participants (customer and supplier) have contributed their capabilities and have shown an absolute willingness to work.

Proof was provided of all guaranteed technical parameters. In accordance with Figure 16, the efficiencies guaranteed by the pump supplier have been maintained or even exceeded. In comparison with the existing pumps, the efficiency has been increased by more than six percent. As a result of the high efficiency, the investment in the replacement of the pumps will be repaid within the planned time.

The short deadlines and the complete scope of work could only be met and progressed in such a safe manner with the care and attention of good specialist personnel and the coordination of the time schedule by the responsible plant engineers.

ACKNOWLEDGEMENT

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NOMENCLATURE

D2	(mm) or (in)	Impeller outside diameter
di	(mm)	Inside diameter of pipe
f	(l/s)	Rotational frequency
h	(kJ/kg)	Enthalpy
L	(mm) or (in)	Bearing span
n	(rev/min)	Speed
NPSH	(m) or (ft)	Net positive suction head
P	(kW)	Power
Q	(m3/h) or (l/s) or (gpm)	Flow
s	(kJ/kg°K)	Entropy
sD	(mm) or (% of nominal clearance)	Diametral clearance
S	(μm) or (mils)	Amplitude (half)
T	(°C) or (°K) or (°F)	Temperature
v	(mm/s) or (in/s)	Vibration velocity

y	(kJ/kg)	Correction factor
Δh is	(kJ/kg)	Differential isentropic enthalpy
η	(%)	Efficiency
ρ	(kg/l)	Specific weight

Subscripts:

d	Discharge
ext	External losses
eff	= RMS (Root Mean Square)
E	Balancing flow
ER	Friction losses of the piston
i	Inside
is	Adiabatic or Isentropic
max	Maximum
mech	Mechanical losses (bearing, seals, convection, radiation, etc.)
min	Minimum flow
k	Behind piston
pol.	Polytropic
R	Friction
RR	Wall friction of impeller (s)
s	Suction
sl	Large diameter wear ring
s3	Small diameter wear ring
3%	3% head drop per stage

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